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ABSTRACT

High pressure twin screw compressors have been widely employed in fuel gas boosting and petrochemical industry. Recently, such compressors, whose maximum of discharging pressure is 5.3MPa, are also adopted in high temperature NH3 heat pumps and NH3/CO2 cascade refrigeration system. However, high pressure twin screw compressors are required to have large capacity, good performance and excellent stability at high operating pressure. In this paper, a semi-empirical model for open-type high pressure twin screw compressor is developed. Experimental research is conducted for identification of parameters, while validation is also made on the accuracy of the model. On the basis of theoretical and experimental research, the performance of the compressor, which includes volumetric efficiency, adiabatic efficiency, discharge temperature and lubricant oil flow rate/temperature, are illustrated. The change pattern of such features on the operating conditions, slide valve loadings and ambient features are then analyzed. Additionally, the stability test results of the high pressure twin screw compressor including the vibration and noise are also shown.

1. INTRODUCTION

With the substitution process of CFCs and HCFCs refrigerant, investigations on better choices for low temperature refrigerant, which is frequently employed in cold storage, instant freezer, pharmaceuticals and food industry, attracts global attention.

Natural refrigerants are proved to be promising in the low temperature refrigeration fields, among which CO2 stands out for its excellent mass and heat transfer properties, as well as large specific refrigeration capacity. CO2 has been widely employed in commercial or industrial refrigeration applications. However, due to the thermodynamic properties of CO2, specific requirement are made for the component of the system, especially the CO2 compressor.
First, the CO2 refrigeration systems are usually working under specially high operating pressure. Therefore, the components of CO2 compressors should has excellent stability while working at extreme operating conditions. The moving parts as bearings or sealing must has enough loading capacity. Secondly, the compressing ration of CO2 refrigeration system is usually low, making the compressor to work under a condition of big pressure difference but low compressing ratio. The internal leakage of compressor appears to the key feature determining the performance of the system. Thus, the sealing capacity of moving components and the better control of leakage paths has direct influence on compressor performance. Additionally, CO2 has large specific refrigeration capacity, which is 5.12 times that of R22, so that the compressor must be made smaller. The manufacturing difficulty for either compressor body but also bearings or sealing are much higher. Moreover, CO2 has high ratio of specific heats, so that the discharge temperature might be high.

The CO2 refrigeration systems are divided into trans-critical and sub-critical systems. In trans-critical systems, the discharged CO2 is beyond critical point so that the refrigerant temperature change within the gas-cooler and the efficiency remains a big problem. On the other hand, the stability and efficiency of compressor is usually low at such extremely high pressure. In recently years, subcritical CO2 refrigeration systems are frequently employed in cascade refrigeration systems, including CO2/NH3, CO2/R22, CO2/R290 systems. The cascade refrigeration systems could have both high stability and efficiency for low temperature refrigeration. Moreover, the nature refrigerant employed in the systems have great effect on the substitution of CFCS/HCFCs refrigerant.

As the core component of refrigeration systems, high pressure CO2 compressors attract much attention these years. The CO2 compressors should has sufficient efficiency and reliability. Twin screw compressors have excellent efficiency and reliability in large capacity systems and high pressure situations, while refrigerant injection contributes to better performance and lower discharge temperature. In CO2 subcritical refrigeration systems, the compressor may works at as high as operating pressure of 5.3 MPa, so that the performance and stability of components should be reevaluated. Conventional approaches for twin screw compressor performance prediction employs either CFD or working process modeling, but such approaches might be too complex and time consuming for applications in practical engineering. Recently, much investigation has been conducted on developing semi-empirical model for twin screws compressors. However, current model are not developed for high pressure twin screw compressors, so that the inter-stage leakage, heat transfer with oil and losses in compressing process are not profoundly considered.

In this paper, a novel high pressure 5.3 MPa CO2 twin screw compressor set is designed and analyzed. Experimental test is conducted on the performance and stability of the compressor. Semi-empirical mathematical model is also built for the prediction of performance, while comparison between simulated and experimental results are compared on the accuracy of prediction for volumetric efficiency, adiabatic efficiency and COP. Their variation with change of operating conditions are also illustrated.

2. SEMI-EMPIRICAL COMPRESSOR MODEL

A series of semi-empirical model for the performance prediction of twin screw compressor and refrigeration system is developed. CO2 is selected as the refrigerant. The condensing temperature ranges from -15 to 15 ℃, while the evaporating temperature ranges from -54 to -15 ℃.

The following assumptions are made to simplify the thermodynamic analysis.
(1) Steady-state assumption. All components are working at steady-state and dynamic characteristics are neglected.
(2) All throttling process are viewed as isenthalpic process.
(3) The heat and pressure loss in pipelines are neglected.

2.1 Model description

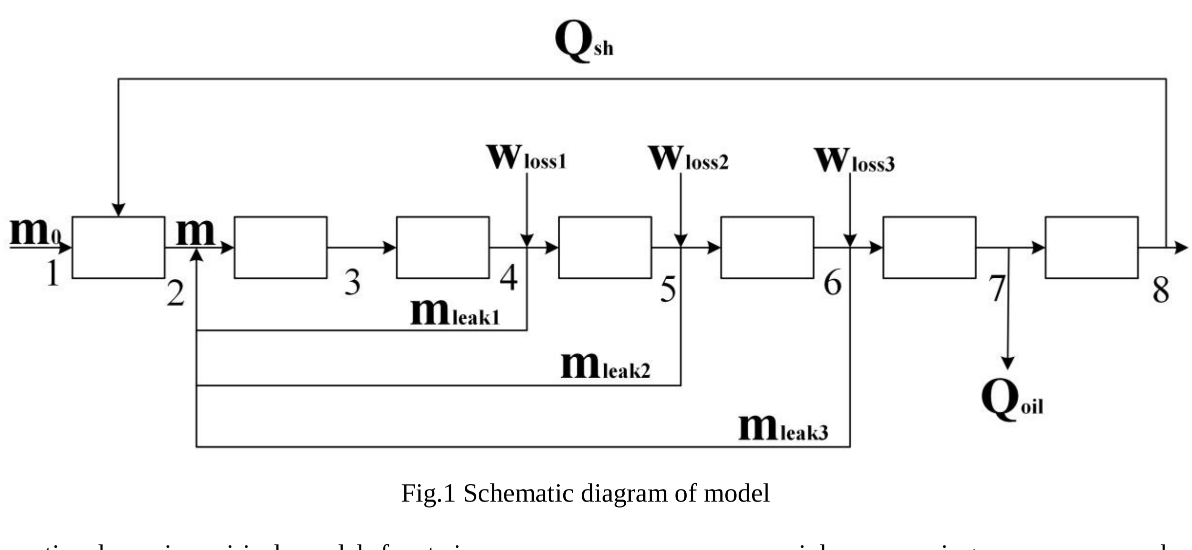

Fig.1 Schematic diagram of model

Conventional semi-empirical model for twin screw compressors are mainly concerning compressors whose operating pressure is ordinary .However, compressors have quite different working process and leakage characteristics at extremely high operating pressure. The inter-chamber leakage for high pressure twin screw compressor could not be neglected as it is in low pressure sets. Since as much as 30-40 percent of refrigerant may leak to suction side at extreme operating conditions, the calculation of power consumption for this part of compressing should be reconsidered. Therefore, in this paper, the compressing process is divided into three individual procedures, and the leakage and power losses is separately calculated after each of the three compressing process. Moreover, the heat exchange between refrigerant and oil for high pressure compressor will be more significant compared with the cooling effect of ambient condition. Therefore, in this paper, the cooling effect of oil, depending on the oil flow rate and oil suction temperature is calculated. Fig.1 illustrates the schematic processing of the semi-empirical model.

The refrigerant suction heating is considered in stage 1-2, while calculated the increased enthalpy of refrigerant contributing to the high temperature of compressor body or rotors. The amount of heat transfer from the rotor and compressor body may depend on the temperature difference between suction and discharge refrigerant as well as the time consumed.

$$Q_{1-2} = f_{1-2}\frac{am}{2\pi n}\left(t_{dis} - t_{suc}\right) \tag{1}$$

The heat transfer coefficient, as well as that between oil and refrigerant introduced later, is formulated as

$$f \propto \left(\frac{T_1 - T_2}{L}\right)^{\frac{1}{4}} \tag{2}$$

Stage 2-3 refers to the refrigerant mixing of refrigerant sucked and refrigerant leaked from higher pressure compressing chamber. The mass of refrigerant entering the compressor and the leakage mass mix altogether, which leads to a slight increase in refrigerant enthalpy after this isobarically mixing process.

$$m_0h_2 + m_{leak1}h_4 + m_{leak2}h_5 + m_{leak3}h_6 = \left(m_0 + m_{leak1} + m_{leak2} + m_{leak3}\right)h_3 \tag{3}$$

Stage 3-4 refers to the first stage of imaginary compressing process. The refrigerant will firstly be isothermally compressed according to the built-in volumetric ratio in this stage. The ratio is calculated based on the relation between inter-rotor volume and rotating angle. Then, part of the refrigerant will be leaked to the suction chamber, after which the viscous friction torque losses are calculated. The compressing of refrigerant is viewed as isentropic.

$$s_3 = s_4 \tag{4}$$

$$v_4 = v_3 / BVR_{3-4} \tag{5}$$

The leakage of refrigerant in twin screw compressors are treated as leakage from leakage paths, which is one unique fictitious path connecting the refrigerant of high and low pressure. In this paper, the leakage mass flow rate is viewed as an isentropic flow through a nozzle.

$$\dot{m}_{leak1} = \frac{b}{v_4}\sqrt{p_4 - p_2} \tag{6}$$

Additionally, the power loss caused by viscous friction torque is calculated. The amount of shaft is consumed as proposed below, and equal amount of heat is viewed as transfer to the refrigerant at this point.

$$P_{loss3-4} = c\mu_{oil}V_g n^2 \tag{7}$$

Stages 4-5 and 5-6 are similarly calculated so that no redundant explanation is given.

Stage 6-7 is considering the discharge process of refrigerant after the compressing process. Either over-compression or under-compression may occur when the discharge port opens. The discharge process would be viewed as isentropic procedure where refrigerant is compressed or expanded by the backpressure refrigerant and then pushed out of the control volume by the rotor.

$$P_{6-7} = v_6 p_{dis} m \tag{8}$$

The heat transfer between oil and refrigerant is considered after the discharge process, because in practical process the time consumed for refrigerant to go through the compressing chamber is far less than enough for the profound heat transferring from high temperature refrigerant to low temperature oil. On the other hand, the heat transfer between compressor body and ambient temperature is neglected.

$$Q_{7-8} = \frac{d}{n}mq_{oil}\left(t_{dis} - t_{oil}\right)^{\frac{5}{4}} \tag{9}$$

Another shaft power loss necessary to be considered should be the transmission efficiency which is caused by the friction in bearings, sealing and gears.

$$P_{loss} = eP_{sp} \tag{10}$$

Therefore, the volumetric efficiency and adiabatic efficiency of the twin screw compressor could be calculated.

$$\eta_v = \frac{m}{m_0} \tag{11}$$

$$\eta_a = \frac{P_{th}}{P} \tag{12}$$

$$P_{th} = m\left(h_{dis}^{ic} - h_{suc}\right) \tag{13}$$

As for the evaporator, the amount of heat supply may depend on the refrigerant flow rate and the enthalpy difference between evaporator inlet and outlet.

$$\dot{Q}_e = \dot{m}_e\left(h_{out} - h_{in}\right) \tag{14}$$

$$COP = \frac{\dot{Q}_c}{P_t} \tag{15}$$

The thermodynamic property of ammonia refrigerant should be accurately calculated in the system modeling and performance prediction. In this paper, all properties as enthalpy, temperature, pressure, entropy are calculated via the NIST Standard Reference Database, which help determine the suction/discharge condition of compressor, and the status of refrigerant at each inlet or outlet of control volume.

2.2 Identification of parameters

As shown in the model description section, totally six parameter are necessarily to be calibrated, including *a* in the calculation of suction heating degree, b_4,b_5,b_6 in the calculation of refrigerant leakage mass flow rate of process 4,5,6, *c* in the calculation of viscous friction power loss and *d* in the calculation of transmission loss. Therefore, six points of experimental results are required in the calibration, where different suction pressure, discharge pressure, suction superheat degree, oil temperature and mass flow rate should be tested in the experiment.

In practical situations, the experiment is conducted on different operating conditions in order to better calibrate the parameters. The evaporating temperature ranges from 20 to 45 ℃ while the condensing temperature ranges from 60 to 85 ℃. The suction superheat degree ranges from 5.91 to 14.63 ℃ so that the suction temperature may be different in spite of constant suction pressure. The oil temperature are precisely controlled thus ranging from 52.7 to 65 ℃ in order to investigate the influence of oil temperature on performance. The oil flow rate ranges from 67 to 236 L/min. Therefore, totally fourteen points of experiment results are employed in this investigation, where six of them are used to calibrate the parameters while the eight points left are used to validate the accuracy.

The MATLAB optimization tools are employed to implement the model of compressor with the error target as follows.

$$err = \sqrt{\sum\left(\frac{\eta_a^{sl}}{\eta_a^{test}} - 1\right)^2 + \sum\left(\frac{\eta_v^{sl}}{\eta_v^{test}} - 1\right)^2 + \sum\left(\frac{t_{dis}^{sl}}{t_{dis}^{test}} - 1\right)^2} \tag{16}$$

3. EXPERIMENTAL FACILITY

In order to examine the performance and stability of high pressure twin screw compressor and CO2 refrigeration system, an experimental plant is built based on the RCH 16S high pressure compressor. The number of tips of the male rotor is 5 while that of the female rotor is 7. The length of rotors are 125mm. The diameter of male and female rotors are 189.5mm and 185mm respectively. Its theoretical discharge volume is 603.84 m^3/h. The schematic diagram is shown in Fig. 2. In application, the condensing temperature ranges from -15 to 15 ℃, and the evaporating temperature ranges from -54 ℃ to -15℃.

Fig.2 Experimental facility

The experiment content includes:

(1) Stability test. The stability of components is examined at high operating pressure, including the stability of bearings, slide valve reliability in loading control and the performance of oil pumps.
(2) Vibration and noise test. The intensity of vibration as well as noise is measured at different directions of the compressor in order to guarantee safe operation. The variations of vibration and noises under different operating conditions are also analyzed.
(3) Capacity test. The mass flow rate of refrigerant is measured at different operating condition, so that the volumetric efficiency could be calculated. At extreme high pressure difference condition, the capacity deterioration could then be calculated.
(4) Power consumption test. According to the measurement of power consumption on motors, the adiabatic efficiency could be calculated. Therefore, the variation of compressor performance, refrigeration system COP and cooling capacity could be analyzed with the change of operating conditions.
(5) Oil cooler test. The load and performance of oil cooler is examined in order to judge whether the requirement of oil cooling is satisfied, and to test the performance of oil cooler at high discharge temperature situation.
(6) Extreme operating condition test. All components are examined on the reliability at extreme operating condition of twin screw compressor.

4. RESULTS AND DICUSSION

The performance evaluation of compressor as well as refrigeration system are made by two steps. First, the volumetric efficiency, isentropic efficiency and COP of heat pump system is calculated based on the experimental data for the power consumption, refrigerant flow rate, heating capacity and operating conditions. Therefore, the parameters should be firstly calculated according to the variation of all parameters with the change of operating conditions, and then the compressor model is capable of predicting the performance of twin screw compressor and refrigeration system. The accuracy of simulated results are shown on volumetric efficiency, isentropic efficiency and COP. Second, with validated accuracy of the model, the refrigeration efficiency, as well as its variation with operating condition, is given.

4.1 Validation of modeling

4.1.1 Volumetric efficiency

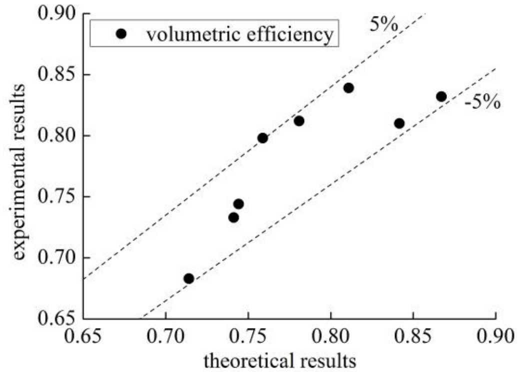

Fig.3 Accuracy of volumetric efficiency

The accuracy of volumetric efficiency prediction is shown in Fig.3. The X coordinate represents the simulated results while the Y coordinate represent the experimental results. At is shown in the figure, the simulated results of volumetric efficiency are within a maximum error of 5% compared with the experimental results in most situations. The volumetric efficiency is prominently higher when the pressure difference between suction and discharge is smaller. The difference of volumetric efficiency at different operating condition could be as high as 0.2. This is mainly the result of leakage which is much more significant when the pressure difference between the suction and discharge sides is larger.

4.1.2 Isentropic efficiency

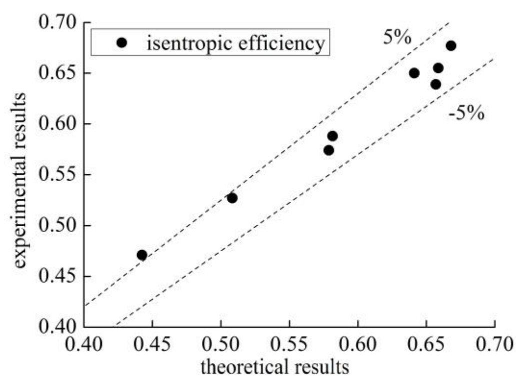

Fig.4 Accuracy of isentropic efficiency

The comparisons of simulated and experimental results of isentropic efficiency are shown in Fig.4. The error of simulated results is also controlled at the maximum of 5% for all operating situation. As it is similar with the trend of volumetric efficiency, the isentropic efficiency is higher when the compressing ratio is lower. The maximum difference at different operating condition is 0.25. The results may attribute to the inter-lobe leakage which is much more significant at bigger pressure difference.

4.1.3 COP

The comparisons of simulated and experimental results of system COP are shown in Fig. 5. The COP of the system ranges from 0.9 to 5 . Such trend is the combined result of theoretical COP increase and a higher isentropic efficiency of compressor at a smaller temperature difference between condensing and evaporating temperature.

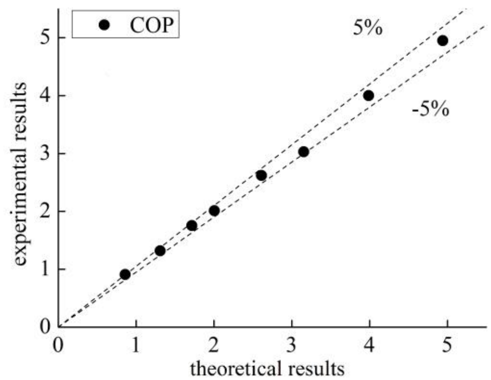

Fig.5 Accuracy of COP

4.2 Variation of performance

4.2.1 Volumetric efficiency

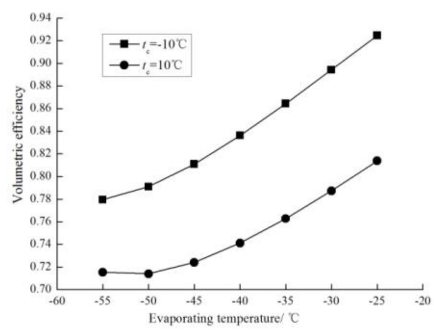

Fig. 6 Influence of evaporating temperature on volumetric efficiency

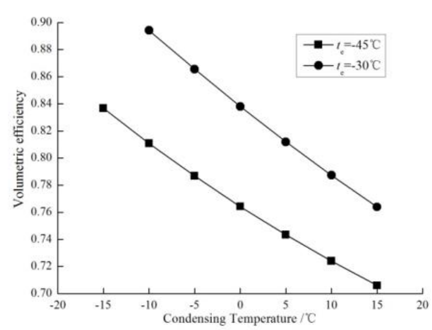

Fig.7 Influence of condensing temperature on volumetric efficiency

As shown in Fig.6 and Fig.7 , the volumetric efficiency decrease significantly with the increase of condensing temperature and the decrease of evaporating temperature. The compressor has 0.2 lower volumetric efficiency when the condensing temperature increase from -15 ℃ to 15℃ and the evaporating temperature decrease from -25℃ to -55℃, which reaches as high as 0.92. This would attribute to the bigger pressure difference between suction and discharge side, as well as more significant suction heating due to higher discharge temperature once the compressing ratio is higher.

4.2.2 Isentropic efficiency

The variations of isentropic efficiency with changing condensing/evaporating temperature are respectively illustrated in Fig.8 and Fig.9 . The adiabatic efficiency will first increase and then decrease with the increase of condensing temperature and the decrease of evaporating temperature. When the refrigerant evaporates at the temperature of -30℃, the adiabatic efficiency reaches its highest of 0.72 at 0℃ of condensing temperature. The efficiency ranges from 0.46 to 0.73 according to the operating condition. Such trend should attribute to the over-compression and owing-compression of screw compressors. If the compressing ratio is lower than the designed operating condition, the compressor will has lower adiabatic efficiency even if the pressure difference is lower.

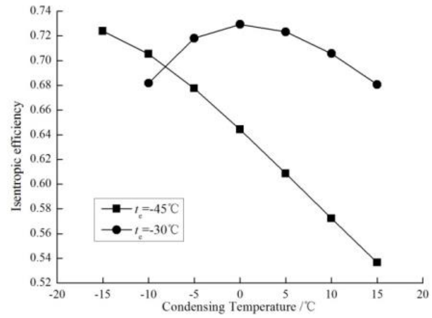

Fig. 8 Influence of condensing temperature on isentropic efficiency

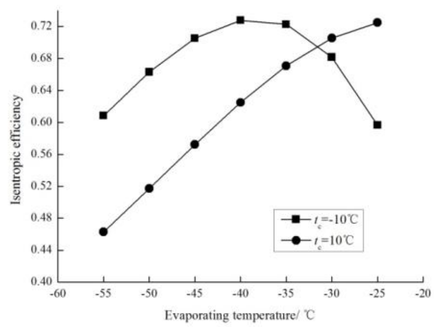

Fig. 9 Influence of evaporating temperature on isentropic efficiency

4.2.3 COP

Fig. 10 and Fig.11 show the variation of COP with changing operating condition. The COP decreases more rapidly when the evaporating temperature is higher or when the condensing temperature is lower. The COP ranges from 0.9 to 6.6 according to the operating condition. The COP difference is approximately 2.2 at 10℃ of condensing temperature, and is bigger with the decrease of heating temperature. Such may attribute to the increase of both the theoretical COP and isentropic efficiency at higher evaporating temperature or lower condensing temperature.

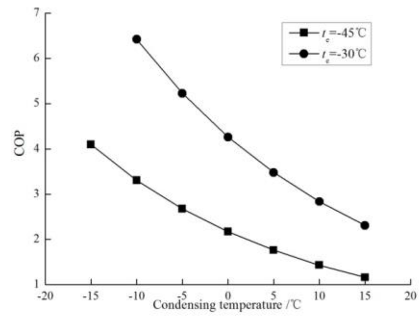

Fig. 10 Influence of condensing temperature on COP

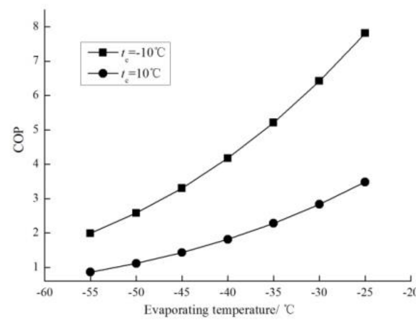

Fig. 11 Influence of evaporating temperature on COP

5. CONCLUSIONS

In this paper, the performance and stability of a high pressure 5.3MPa twin screw compressor is theoretically and experimentally analyzed. Novel semi-empirical model is developed to predict the performance of the CO2 subcritical refrigeration system, which has taken the inter-chamber leakage losses and cooling effect of oil into consideration. An experimental plant is set up to test both the performance of high pressure CO2 twin screw compressor and that of the subcritical refrigeration system. From the validation of compressor efficiency and COP of the refrigeration system, the accuracy of the model could be guaranteed, whose error is mainly controlled with 5%.

The high pressure CO2 twin screw compressor is guaranteed stability and reliability working at even extreme operating conditions. The COP decreases with the increase of condensing temperature, and decreases more rapidly when the evaporating temperature is higher. The COP ranges from 0.9 to 6.6 according to the operating condition. The adiabatic efficiency will first increase and then decrease while the volumetric efficiency may monotonously increase with the increase of condensing temperature and the decrease of evaporating temperature.

NOMENCLATURE

Q	quantity of heat	(kJ)
m	mass	(kg)
t	temperature	(K)
n	rotation speed	(r/min)
L	length	(m)

h	enthalpy	(kJ/kg)
s	entropy	(kJ/(kg·K))
v	specific volume	(m^3/kg)
p	pressure	(kPa)
P	power	(kW)
μ	viscosity	(Pa s)
V	volume	(m^3)
q	mass flow rate	(kg/s)
η	efficiency	(-)
err	error	(-)

Subscript

suc	suction
dis	discharge
sp	shaft power
th	theoretical
ic	isentropic compressing
a	isentropic
v	volumetric
sl	simulate
t	technical
e	evaporator
c	condenser

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